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DOE/NASA/50112-72
NASA TM-100256

Hot Piston Ring Tests

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(NASA-TM-100256) HOT PISTON RING TESTS N88-14486
Final Report (NASA) 17 p CSCL 10B

Unclas
G3/44 0114366

Work performed for
U.S. DEPARTMENT OF ENERGY
Conservation and Renewable Energy
Office of Vehicle and Engine R&D

Prepared for
Twenty-fifth Automotive Technology Development
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E-3808

SUMMARY

As part of the DOE/NASA Automotive Stirling Engine Project, tests were made at NASA Lewis Research Center to determine whether appendix gap losses could be reduced and Stirling engine performance increased by installing an additional piston ring near the top of each piston dome. An MTI-designed upgraded Mod I Automotive Stirling Engine was used for these tests. Unlike the conventional rings at the bottom of the piston, these rings operated in a high temperature environment (700 °C). Because of this, we called them "hot-rings." It was necessary that they be made of a high temperature alloy (Stellite 6B) and that a high temperature solid lubricant coating (NASA Lewis-developed PS-200) be applied to the cylinder walls. Engine tests were run at 5, 10, and 15 MPa operating pressure over a range of operating speeds. Tests were run both with the hot-rings in place and without them to provide a baseline for comparison. Although budget and schedule restrictions severely limited the testing, the minimum data to assess the potential of both the hot-rings and high temperature low friction coating was obtained.

Results indicated a slight increase in power and efficiency with the hot-rings over the baseline configuration. This increase was over and above the friction loss introduced by the hot-rings. Seal leakage measurements showed a significant reduction in leakage with the hot-rings in place. In addition, cylinder wall temperature measurements indicated less cylinder heating in the appendix gap area--between the lower piston rings and the hot-ring. Approximately 22.4 hours of ring-on-coating operation were recorded. After the initial break-in period, wear on both the rings and the coating was low. The PS-200 coating seems to offer significant potential for long-term operation at high temperatures.

Both the hot-ring and the PS-200 low friction coating show promise and should be pursued further. Potential benefits are applicable not only to the Stirling engine, but to the adiabatic diesel as well.

INTRODUCTION

Advanced component technology testing was done at NASA Lewis on a Mod I kinematic Stirling engine this past year. This research and development program was initiated in an attempt to reduce the "appendix gap" losses defined analytically for the Stirling engine. The appendix gap is the volume between

the cylinder wall and the actual piston dome itself as shown in figure 1. This volume is the cross-hatched area. These appendix gap losses in theory result in a decrease in engine performance, namely engine brake power and brake efficiency. A possible way to reduce this appendix gap loss is to install an additional piston ring at the top of the piston dome to eliminate or reduce working gas flow in and out of the appendix gap. The conventional piston configuration is shown at the left of figure 1 with the hot-ring modified version at the right. This cannot be done with the standard Rulon ring because of the extremely harsh environment that this ring must withstand in this region. The operating temperature in this region is approximately 700 °C (1292 °F) with a hydrogen atmosphere. A high temperature metal alloy is needed to survive under these conditions. Stellite 6B was used for these tests. Some form of lubrication is also required to allow adequate friction and wear resistance for this hot-ring. A high temperature dry lubricant developed at NASA Lewis was adapted for this purpose.

TEST APPARATUS AND PROCEDURES

The engine used for this testing was a Mod I Automotive Stirling Engine with the auxiliaries, except for the oil pump and hydrogen compressor, removed. Figure 2 is a photograph showing the engine in the test cell on the test stand. The lines running throughout the photo were for research instrumentation.

Figure 3 is a schematic of the engine. The reason for introducing this schematic is to show where the modified components are located with respect to the rest of the engine components. Modified piston domes were attached to the standard piston bases and modified cylinder heads were installed in place of the original cylinder heads.

Special Materials Requirements

As mentioned previously, some form of dry high temperature lubricant was required to allow operation of the hot-ring in the high temperature cylinder environment. A material developed at NASA Lewis (PS-200) looked promising. PS-200 has a base of 48 percent chrome carbide and 32 percent nickel aluminide with 10 percent calcium and barium fluoride eutectic, and 10 percent silver. The chrome carbide-nickel aluminide provides the wear resistance, while the eutectic-silver mix provides the high temperature lubricant. Pin-on-disk wear and friction tests were made in both helium and hydrogen at temperatures up to 760 °C (refs. 1 and 2). Several potential ring materials were tried initially before the Stellite 6B was selected. Figure 4 shows the results of these tests. PS-200 shows a significant reduction in friction coefficient over the chrome carbide--from 0.6 to less than 0.2. Wear rates were low. These tests gave us confidence that we could run the hot-ring equipped engine stably for a long enough time to provide the data necessary for performance comparison.

Test Hardware

The piston ring configurations shown in figure 5 were used for comparison purposes during the engine testing. The hot-ring configuration on the left side was designed by Dr. Roy Howarth of MTI. This consisted of an upper and a lower half to the piston dome which allowed for the removal or replacement of

the additional piston ring. The actual design of the hot-ring configuration included an upper wave spring to keep the hot-ring seated against the lower half of the piston ring groove, an expander ring behind the hot-ring to apply uniform pressure against the cylinder bore, and the hot-ring itself. Standard split-solid Rulon rings were used in the conventional piston ring base grooves. Note the vent holes in the piston base which allowed for manifolding of the minimum cycle pressures in each of the four cylinders. The configuration on the right is a schematic of the baseline piston dome configuration employed for the purpose of comparison. A filler ring was mounted in place of the hot-ring to allow for a uniform transition between the upper and lower half of the piston dome.

Figure 6 is a photograph of the piston dome in the hot-ring configuration after approximately 22.4 hours of testing. There is significantly less discoloration associated with this hot-ring configuration than with the conventional arrangement. This apparent thermal dam effect will be discussed later.

Figure 7 is a photograph of a high temperature dry lubricant coated cylinder. As you can see, the darker portion of the photo is where the additional piston ring rode along the stroke of the bore and wore in the coating. The lighter portion is the original diamond-ground, flame sprayed PS-200 coating surface. This coating was flame sprayed and then ground to the particular specifications needed to meet design requirements. Actual measurements from inspection showed that there was only about 0.0002 in. wear associated due to the ring-on-coating contact after 22.4 hours of testing. The surface finish measurement for the as-ground coating was 80 μ-in. RMS. After wear-in, the surface finish was a very smooth 5 μ-in. It would appear that almost half the apparent wear was simply reduction of the original surface asperities. It is not clear whether the actual steady-state wear rates would be acceptable for real engine use. However, our purpose was well served.

Friction

It was necessary to account in some way for the added friction of the hot-ring in order to assess what actual thermodynamic benefit may have resulted. Ring bearing pressures were calculated based on assumed pressure differences across the ring over the cycle. These were integrated along with the piston speed to provide an average value of PV for each operating pressure and engine speed. The total ring surface area was assumed to be the friction bearing surface used in the friction loss calculations. Figure 8 shows friction power loss in kW/cycle as a function of engine speed. At the full-power point of 15 MPa working pressure and 4000 rpm, the total engine loss associated with the hot-ring friction is approximately 2.5 kW and decreases linearly with speed to zero.

RESULTS AND DISCUSSION

The total working gas leakage past the piston rings was manifolded together (as mentioned previously) and measured in an attempt to gauge the sealing effectiveness of the hot-rings. Leakage normally occurs up across the bottom ring as well as down across the top ring. The hot-ring, of course, is intended to restrict only the flow from the top down. The lower ring is expected to leak similarly in either configuration. A reduction in overall

leakage implies a substantially greater reduction in the flow from the top only. Figure 9 shows a comparison of the leakage between the baseline and hot-ring configurations at 5 MPa operating pressure. A significant reduction is seen. This implies that the hot-ring seals very well and does prevent flow in and out of the appendix gap. Similar results were obtained at 10 and 15 MPa.

A further corroboration of the sealing effect was obtained by measuring the cylinder external wall temperatures. Figure 10 shows typical results of these measurements. These data were taken at 5 MPa and 1000 rpm. The cylinder schematic shows the thermocouple locations. Temperatures for the hot-ring configuration are consistently lower down the cylinder indicating less hot gas flow into the appendix gap.

In order to obtain a direct comparison of hot-ring performance with the standard configuration, tests were run at 5, 10, and 15 MPa over a range of engine speeds. Figure 11 shows brake power as a function of engine speeds for both configurations at all three operating pressures. Unfortunately, a malfunctioning air/fuel control system limited the maximum fuel flow available and operation was not feasible beyond about 3500 rpm at 10 MPa and 2000 rpm at 15 MPa. The hot-ring power level appears to be slightly higher (as much as 6 percent) at all operating conditions. We had hoped for more. However, we must also consider that this apparent small net gain was achieved even though total ring friction was increased with the added hot-ring. If this is accounted for, the apparent thermodynamic gain due to appendix loss reduction alone was up as much as 11 percent in power.

Figures 12 to 14 show an efficiency comparison for the same tests. Again, as with the power, efficiency appears slightly higher for all operating conditions. Taking the added friction into account gives essentially the same results as for power--maximum apparent thermodynamic gains of 11 percent or more.

CONCLUDING REMARKS

Although not conclusive, these data indicate that appendix gap losses are real (in some form) and may be reduced if flow in and out of the gap is reduced. On the assumption that this initial effort did not produce an optimum technique to implement this reduction, more effort in this direction would appear to be warranted--not only to improve engine performance, but also to define more clearly a significant loss mechanism for input to the Stirling engine simulation. A side benefit of this work was the first use in a real engine of a dry ring lubricant. The PS-200 coating not only made our limited testing possible, but also appears to offer definite promise for practical engine use. It may even be applicable to the adiabatic diesel where high temperature ring lubrication has been a prime deterrent to effective application.

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1. H.E. Sliney, "A New Chromium Carbide-Based Tribological Coating for Use to 900 °C with Particular Reference to the Stirling Engine," Journal of Vacuum Science and Technology, Vol. 4, No. 6, Nov.-Dec. 1986, pp. 2629-2632.

2. C. Della Corte and H.E. Sliney, "Composition Optimization of Self-Lubricating Chromium-Carbide-Based Composite Coatings for Use to 760 °C," ASLE Transactions, Vol. 30, No. 1, Jan. 1987, pp. 77-83.

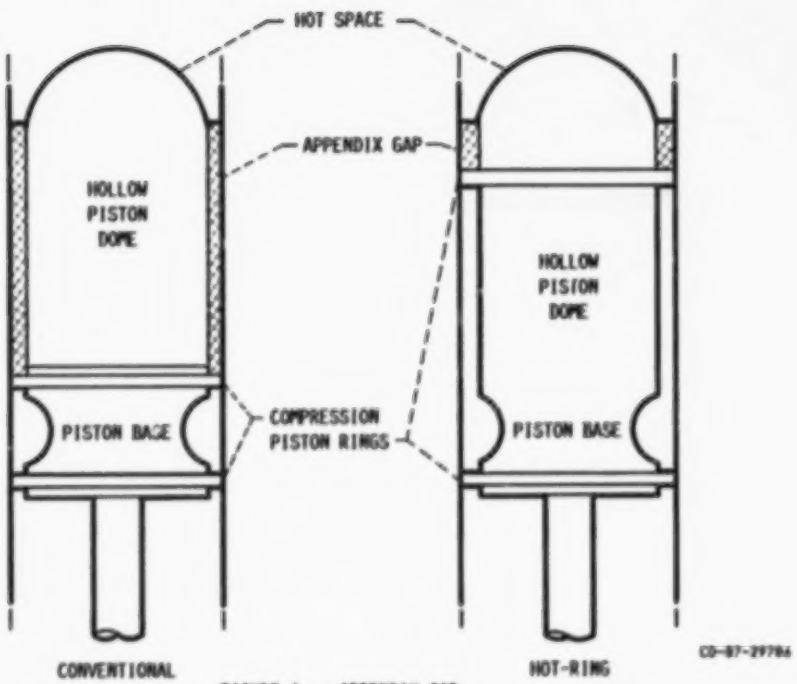

FIGURE 1. - APPENDIX GAP.

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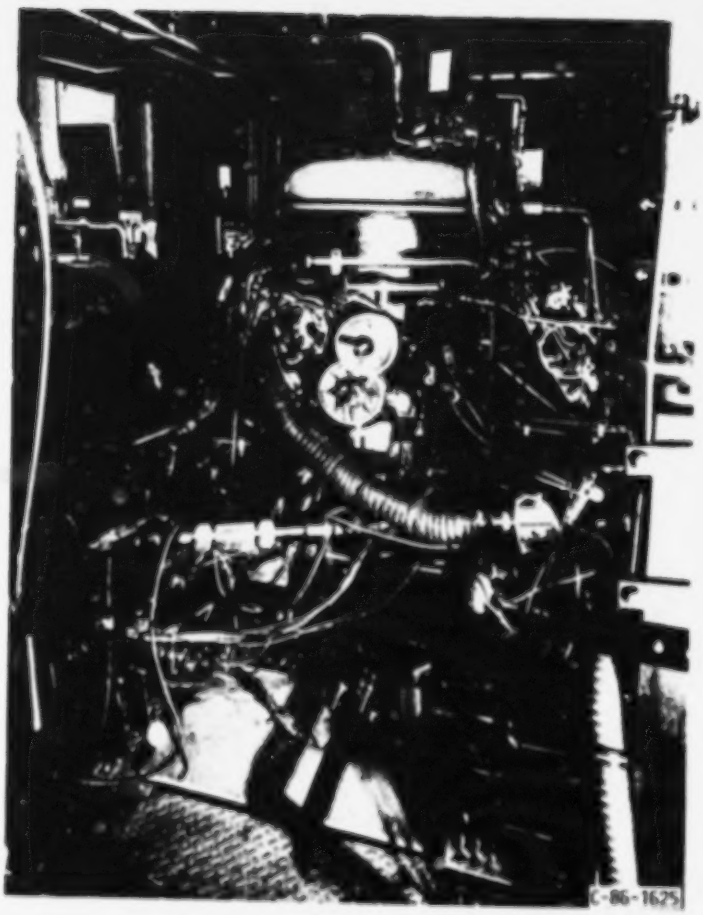

FIGURE 2. - ENGINE IN TEST CELL.

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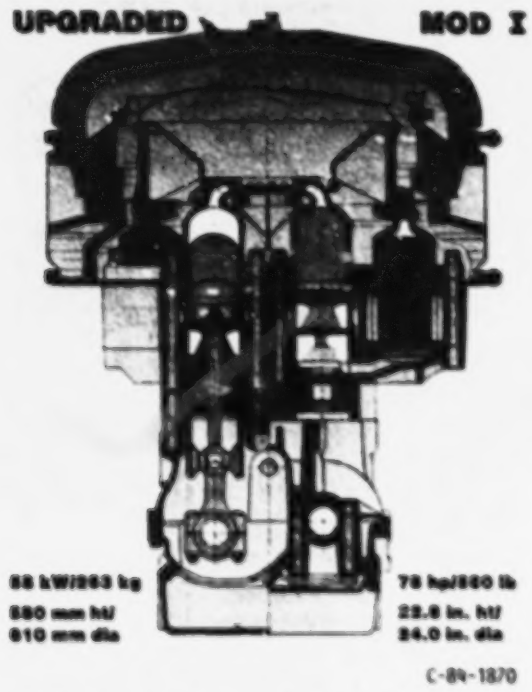

FIGURE 3. - MOD I AUTOMOTIVE STIRLING ENGINE.

FRICTION OF STELLITE 6B SLIDING ON BONDED CHROMIUM CARBIDE AND ON PS200

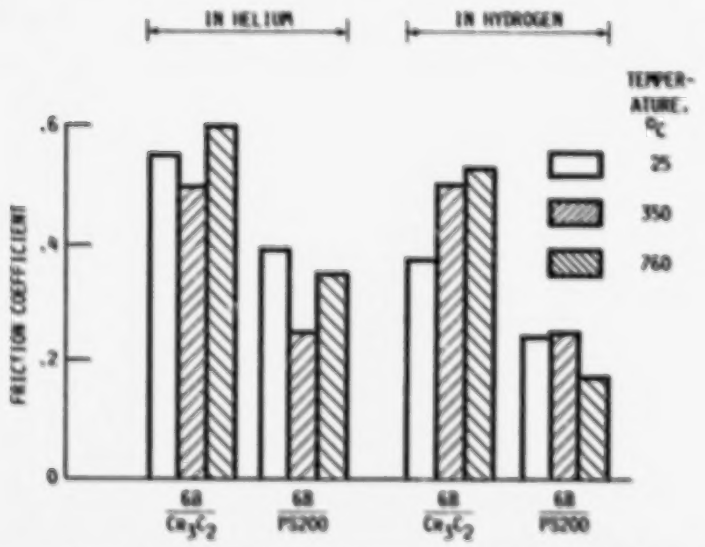

FIGURE 4. - BONDED CHROMIUM CARBIDE AND PS 200 IN STIRLING ENGINE ATMOSPHERES.

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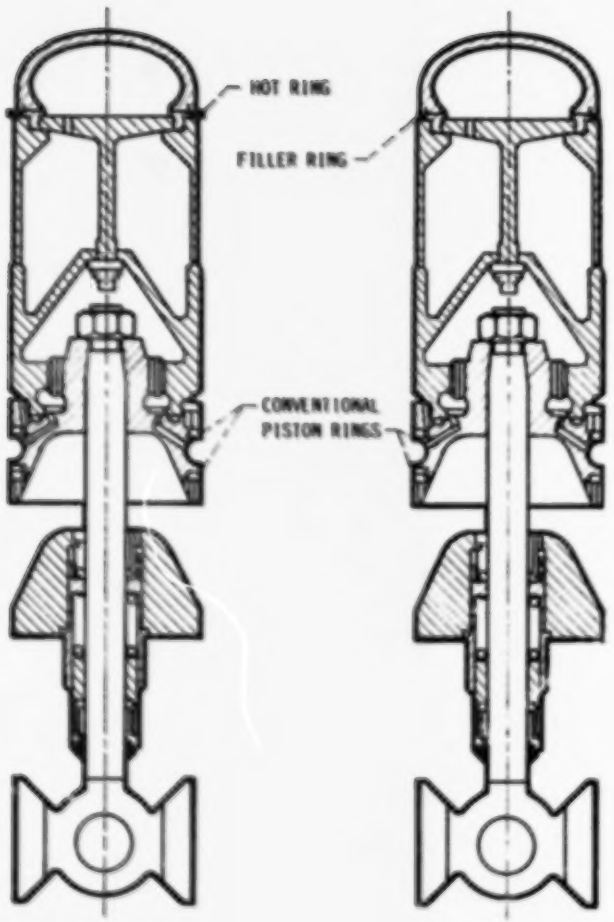

FIGURE 5. - PISTON CONFIGURATIONS. CD-87-29187

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FIGURE 6. - HOT-RING PISTON.

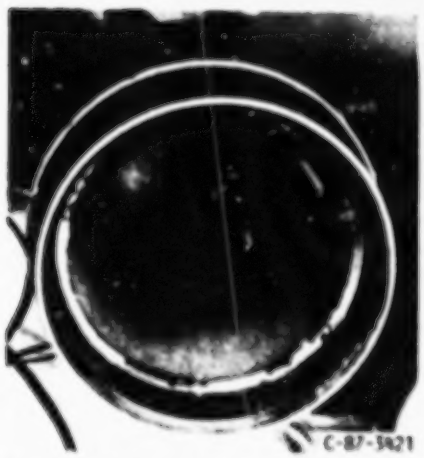

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FIGURE 7. - COATED CYLINDER AFTER 22.4 HOURS OF OPERATION WITH HOT-RINGS.

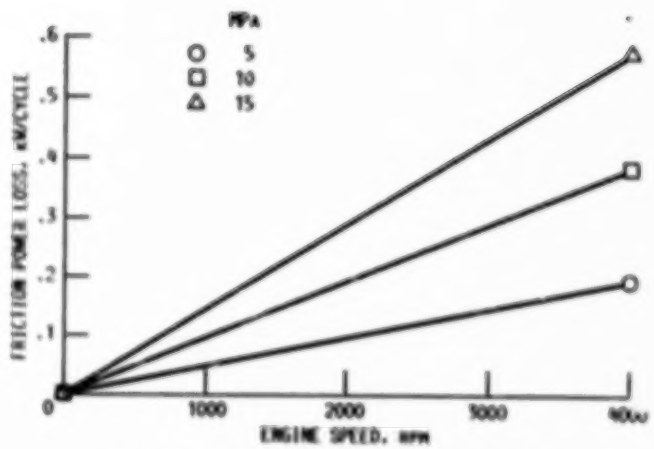

FIGURE 8. - HOT-RING FRICTION LOSS (CALCULATED).

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MOD I STIRLING ENGINE: 5 MPa MEAN PRESSURE

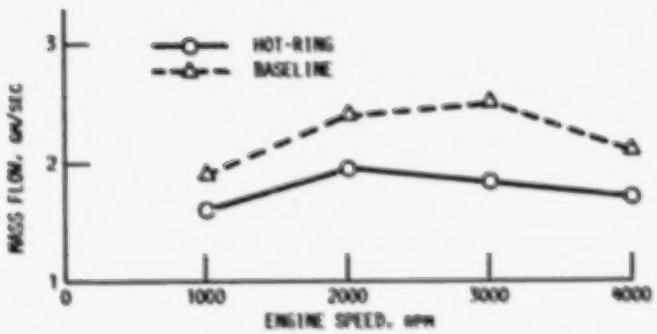

FIGURE 9. - SEAL LEAKAGE COMPARISON.

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MOD I STIRLING ENGINE: 5 MPa MEAN PRESSURE: 1000 RPM

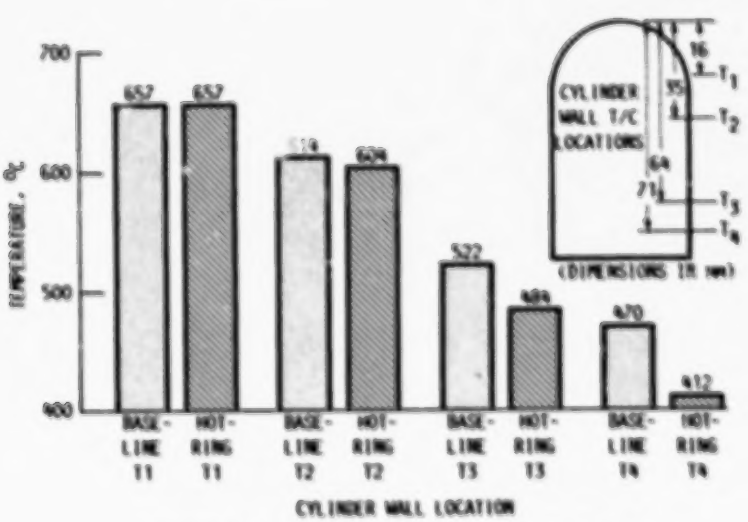

FIGURE 10. - CYLINDER WALL TEMPERATURE COMPARISON.

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MOD I STIRLING ENGINE

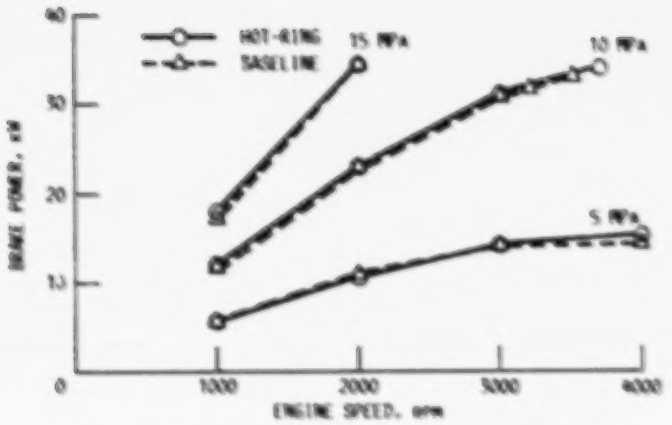

FIGURE 11. - POWER COMPARISON-HOT-RING VERSUS BASELINE.

MOD I STIRLING ENGINE: 5 MPa MEAN PRESSURE

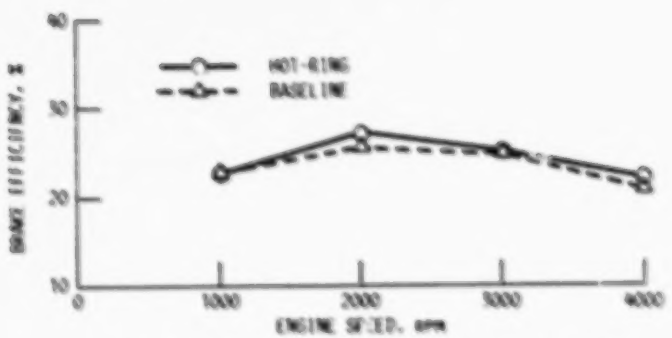

FIGURE 12. - EFFICIENCY COMPARISON-HOT-RING VERSUS BASELINE.

MOD I STIRLING ENGINE; 10 MPa MEAN PRESSURE

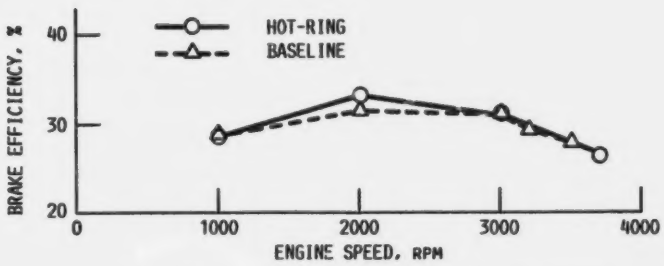

FIGURE 13. - EFFICIENCY COMPARISON-HOT-RING VERSUS BASELINE.

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MOD I STIRLING ENGINE; 15 MPa MEAN PRESSURE

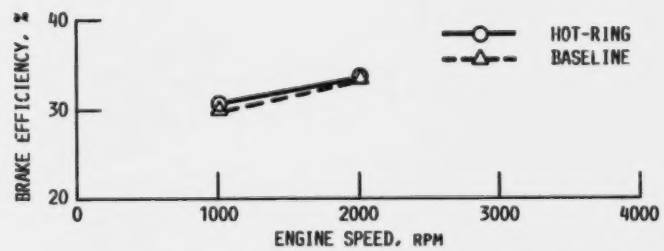

FIGURE 14. - EFFICIENCY COMPARISON-HOT-RING VERSUS BASELINE.

CD-87-29792

NASA
National Aeronautics and Space Administration

Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA TM-100256		

4. Title and Subtitle	5. Report Date
Hot Piston Ring Tests	December 1987
	6. Performing Organization Code

7. Author(s)	8. Performing Organization Report No.
David J. Allen and William A. Tomazic	E-3808
	10. Work Unit No. 778-35-13

9. Performing Organization Name and Address	11. Contract or Grant No.
National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135-3191	
	13. Type of Report and Period Covered: Technical Memorandum

12. Sponsoring Agency Name and Address	14. Sponsoring Agency Code
U.S. Department of Energy Office of Vehicle and Engine R&D Washington, D.C. 20545	DOE/NASA/50112-72

15. Supplementary Notes

Final Report. Prepared under Interagency Agreement DE-AI01-85CE50112. David J. Allen, Sverdrup Technology, Inc., Middleburg Heights, Ohio and William A. Tomazic, Lewis Research Center. Prepared for the Twenty-fifth Automotive Technology Development Contractors' Coordination Meeting, sponsored by the U.S. Department of Energy, Dearborn, Michigan, October 26, 1987.

16. Abstract

As part of the DOE/NASA Automotive Stirling Engine Project, tests were made at NASA Lewis Research Center to determine whether appendix gap losses could be reduced and Stirling engine performance increased by installing an additional piston ring near the top of each piston dome. An MTI-designed upgraded Mod I Automotive Stirling Engine was used for these tests. Unlike the conventional rings at the bottom of the piston, these rings operated in a high temperature environment (700 °C). Because of this, we called them "hot-rings." It was necessary that they be made of a high temperature alloy (Stellite 6B) and that a high temperature solid lubricant coating (NASA Lewis-developed PS-200) be applied to the cylinder walls. Engine tests were run at 5, 10, and 15 MPa operating pressure over a range of operating speeds. Tests were run both with the hot-rings in place and without them to provide a baseline for comparison. Although budget and schedule restrictions severely limited the testing, the minimum data to assess the potential of both the hot-rings and high temperature low friction coating was obtained. Results indicated a slight increase in power and efficiency with the hot-rings over the baseline configuration. This increase was over and above the friction loss introduced by the hot-rings. Seal leakage measurements showed a significant reduction in leakage with the hot-rings in place. In addition, cylinder wall temperature measurements indicated less cylinder heating in the appendix gap area--between the lower piston rings and the hot-ring. Approximately 22.4 hours of ring-on-coating operation were recorded. After the initial break-in period, wear on both the rings and the coating was low. The PS-200 coating seems to offer significant potential for long-term operation at high temperatures. Both the hot-ring and the PS-200 low friction coating show promise and should be pursued further. Potential benefits are applicable not only to the Stirling engine, but to the adiabatic diesel as well.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Stirling engine; Appendix gap; Hot piston ring	Unclassified - Unlimited Subject Category 44 DOE Category UC-96

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No of pages	22. Price*
Unclassified	Unclassified		A02

NASA FORM 1626 OCT 86

*For sale by the National Technical Information Service, Springfield, Virginia 22161